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DESIGN, ANALYSIS AND MANUFACTURING OF EPICYCLIC GEARBOX FOR ALL-TERRAIN VEHICLE (ATV)

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ABSTRACT

All-Terrain Vehicles (ATVs) are used to handle a wider variety of terrains and are specifically used in border patrol, oil exploration and surveying. In this study, an Epicyclic Gearbox has been designed, analysed and manufactured for its use in an ATV. Epicyclic Gearbox which is generally used in Tractors, High Speed Gas turbines and Construction Equipment has features such as rotating axes of certain gears, certain gears in mesh with multiple gears at the same time and co-axiality of input and output shafts. This provides for compactness of size, weight reduction and sufficient gear reduction which are essential requirements for ATVs. The modelling was done in Solid Works and the analysis in ANSYS. The gearbox was tested on the ATV, with its usage being effective.

***KEYWORDS:** Epicyclic Gearbox, Power Train, Gearbox Manufacturing, ANSYS, Solid Works*

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Original Article

INTRODUCTION

All-Terrain Vehicles require sufficient amount of torque for them to overcome obstacles in their paths and also to climb steep gradients. The torque from the engine is multiplied and provided to the wheels by means of the transmission. In order to reduce the size and weight of the power train assembly, an Epicyclic gearbox is used.

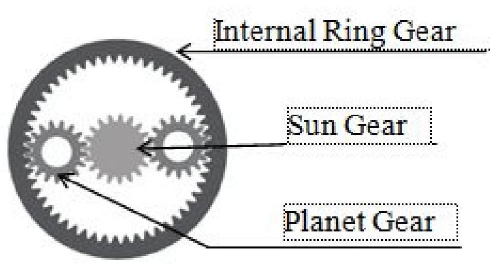


Figure 1: Epicyclic Gearbox Components

The Epicyclic gearbox designed consists of a planetary arrangement in which there is a centrally located sun gear, an internal ring gear, number of planet gears in contact with the sun gear as well as the ring gear and a planet carrier. The planets are mounted on the planet carrier which supports and keeps them together. In this arrangement, one element out of the sun gear, ring gear and the planet carrier is fixed and out of the remaining two, input is given to one and output is taken from the other. The meshing of one gear with multiple gears at the same time leads to the tangential force getting divided at multiple teeth and consequently the force at one tooth is considerably lower. Because of the rotation of the planet gears about their own axis as well as the central axis, large gear reduction can be obtained with low weight, compact assembly and coaxial input and output shafts.

The gearbox has been designed for an international level competition called BAJA. It is organised by SAE (Society of Automotive Engineers) wherein a group of students have to design, manufacture and race an All-Terrain Vehicle (ATV).

Since the teeth of planet gears have to be correctly meshed with both the internal ring gear and the sun gear at the correct centre distance, the following relation governs their number of teeth of each:

$$N_R = N_S + 2\ x\ N_P$$

Where,

N_R = Number of Teeth on Ring Gear

N_S = Number of Teeth on Sun Gear

N_P = Number of Teeth on Planet Gears

The reduction can be calculated as follows:

Table 1: Reduction Calculation

Sr. No	Motion	Carrier	Sun	Planet	Ring
1.	Carrier fixed and +1 revolution given to sun gear	0	1	$-N_S/N_P$	$-N_S/N_R$
2.	Multiply by +x	0	+x	$-x(N_S/N_P)$	$-x(N_S/N_R)$
3.	Add +y revolutions to all elements	+y	+y	+y	+y
4.	Total motion (Adding steps 2 and 3)	+y	x+y	$y-x(N_S/N_P)$	$y-x(N_S/N_R)$

By giving the desired input to suitable element, fixing another suitable element and solving for x and y, the gear reduction can be calculated. Highest gear reduction in the same direction can be obtained when the ring gear is fixed, sun gear is given the input and output is taken from the carrier. This gear reduction is as follows:

$$Gear\ Reduction = 1 + \frac{N_R}{Ns}$$

LITERATURE REVIEW

For design of a gear box, it is necessary to look into the design aspects and literature available in order to better understand the designing aspects. While designing a gearbox, the gears have to be designed to withstand failure due to Wear (i.e. removal of metal from the surface or formation of cracks or pits on the gear surface) as well as bending (insufficient strength of the gear teeth). Bending strength has to take care of the torque being transmitted and can be improved by using material of higher strength, increasing number of teeth, module or face-width. Wear strength has to take care of the high RPM and can be improved by increasing hardness of the teeth or by increasing face-width. When made of the same material, the smallest gear is the weakest and has to be designed to resist these failures. The gears are designed according to Buckingham's Equation for dynamic load and effective load as stated in Design of Machine Elements, V.B Bhandari. While doing this calculation it is necessary to take into account the Prime mover and the duration for which it will be run each day, which will decide the application factor. Another consideration is the machinery which the gearbox will run. This will decide the type of loads (i.e. Uniform, Moderate or Heavy shocks), which account for the Load factor. The method of manufacturing greatly affects the performance of gears. Various methods such as casting, hobbing, shaping,

and milling are possible and thus decide the grade of the gears. Higher the grade, lower is the pitch error, lowering the dynamic forces on the gears. While designing, it is necessary to decide whether the gears will be integral with the shaft or will they be rigidly mounted on the shaft. One piece shaft and gear is generally stronger than separate gear and shaft. But separation provides ease of assembly and manufacturing. Holes if made have to follow certain conditions so that the gear is not weakened considerably. These geometrical relations are followed according to Handbook of Gear Design, Gitin Maitra. Gear material is an important parameter while designing gears. Another consideration is the type of meshing, whether sequential or simultaneous as stated in Handbook of Gear Technology, D.D Chawathe. In Simultaneous meshing, each planet gear is in mesh with the sun at the same point on the tooth flank. Contact at the same point of tooth flank is achieved in sequence for different planets in Sequential Meshing. This results in lower load impulses and less harmful vibrations, which is desirable. Various alloy steels like EN24 as well as aluminium alloys like Aluminium 7075 can be used. While selecting the material, its feasibility for machining and heat treatment should also be taken into account. Case hardening is preferred since it results in hard, wear resistant surface and tough, shock resistant core. Various methods like Carburising, Nitriding, Carbo-Nitriding and Induction Hardening are possible. Some like Nitriding are preferred since quenching is not required in them, thus preventing distortion. Material selection, heat treatment processes were followed according to Material Science and Metallurgy for Engineers, VD Kodgire.

Casing is an important component of any gearbox. Its function is to support the moving components in the gearbox by virtue of bearings. It also acts as a fluid tight container to hold the oil for lubrication. It is the casing, which mounts the gearbox on the chassis of the vehicle. Simplicity of mounting and strength should be given priority while deciding location of mounting points. The number of bolting points for joining the two casing halves were decided according to PSG Design Data-book. Aluminium has been used extensively as a casing material due to its low weight, though other materials can also be used. Provisions for adding and removing oil without disassembly also have to be made. Various methods of manufacturing such as casting, forging and machining are available. While Casting is economical, machining provides accurate dimensional tolerances and forging provides higher strengths. The fixing of the ring gear in the casing has been done by press-fit according to Shigley’s Mechanical Engineering Design. The required amount of interference is calculated to apply the holding torque. In case of Epicyclic Gearbox, a system of Planet Carrier has to be developed. Finite Element Analysis is an effective tool which can be extensively used to decide and judge the safety of various components such as Casing (also to decide number and position of mounting points), Planet Carrier and Pins.

DESIGN CONSTRAINTS

The gearbox has been designed for an All-Terrain Vehicle (ATV) which is run by an Engine with following specifications:

- **Capacity:** 305cc
- **Maximum Torque:** 18.85Nm @ 2600 rpm
- **Maximum Power:** 10Hp @ 3800 rpm.

A CVT (Continuously Variable Transmission), having reduction range from 3.85 to 0.9 is coupled to the engine, just before the Gearbox thus multiplying the input torque to the gearbox. According to CVT reduction range and further transmission layout, the required reduction should be in the range from 3.7 to 4. Due to constraints on the size, it should have maximum box size of 260mm*200mm*105mm and simple mounting to facilitate easy assembly.

Design of Components

The various parts designed for the gearbox are as follows:

- Sun Gear
- Planet Gears
- Ring Gear
- Planet Carrier
- Planet Carrier Pins
- Gearbox Casing along with mounting

DESIGN OF GEARS

Since the CVT connects the Engine to the Gearbox, the gearbox has a maximum input torque of 73 Nm at 676 rpm and maximum power of 10 HP at 4223 rpm. To obtain maximum reduction in same direction, the ring gear is fixed and the sun gear is given the input. The number of teeth on sun, planets and ring are taken as 19, 17 and 53 respectively since a reduction of 3.789 is achieved which is as desired. These numbers satisfy the following relation which ensures proper meshing since one gear is in mesh with multiple gears at the same time:

$$(N_S + N_R) \Big/ Number\ of\ planets = Integer$$

All the gears are taken to be having 20^0 full depth involute profiles, 2mm module (considering availability of cutters) and face width 30 mm. The number of planets has been chosen as three to have an optimum combination of reduced tangential force as well as easy assembly and disassembly along with weight reduction. The planets are placed symmetrically about the sun thus improving the stability of the system. The number of teeth on all the gears are Co-Prime with each other thereby distributing wear uniformly within all the teeth of the gear thus preventing stress concentration. These numbers result in desirable sequential meshing since Number of Teeth on the Sun is not divisible by number of planets. Also, this configuration is in harmony with the size constraint. The material selected is low carbon steel EN353 having yield strength of 750MPa and tensile strength of 930MPa since it can be case hardened by carburising to 600HB.

Gears are designed according to Buckingham's Equation for sufficient strength for bending stress as well as for wear failure. The calculation is done as follows:

- $F_t = 2\ x\ \frac{T}{D_G}\ x\ \frac{1}{3}$ (since three planets are present)

  Where,

  F_t = Tangential Force on the Gear (N)

  T = Torque on the Gear (N-mm)

  D_G = Pitch Circle Diameter of the Gear (mm)

- $F_b = \sigma_b \; x \; m \; x \; b \; x \; Y$

  Where,

  F_b= Beam Strength of Gear Tooth (N)

  σ_b= Bending Endurance Strength (N/mm^2)

  m = Module of the Gear (mm)

  Y = Lewis form factor

  b = Face width of the Gear (mm)

- $\sigma_b = {S_{ut}}/{3}$

  Where,

  S_{ut} = Ultimate Tensile Strength of the Material

- $Y = 0.484 - \frac{2.87}{Z_p}$

Where,

Z_p= Number of teeth on pinion

- $F_w = d \; x \; b \; x \; Q \; x \; k$

  Where,

  F_w= Wear Strength of Gear Tooth (N)

  Q = Ratio factor

  K = Load Stress factor

- $Q = 2 \; x \frac{Z_G}{Z_G + Z_P}$

  Where,

  Z_G = Number of teeth on Gear

  Z_P = Number of teeth on Pinion

- $K = 0.157 \; x \; [\frac{BHN}{100}]^2$

  Where,

  BHN = Brinell hardness of the Pinion

- $Ft_{max} = K_a \; x \; K_m \; x \; F_t$

  Where,

  Ft_{max} = Maximum Tangential Force

K_a= Service Factor

K_m = Load Distribution Factor

- $F_d = \frac{21\ x\ V\ (bC+Ft_{max})}{21\ x\ V+\ \sqrt[2]{bC+Ft_{max}}}$

Where,

F_d= Dynamic Load (N)

V = Pitch line velocity (m/s)

C = Deformation factor (N/mm) (decided according to IS grade of the gear)

- $F_{eff} = Ft_{max} + F_d$

Where,

F_{eff} = Effective load (N)

- For the design to be safe,

$F_d > F_{eff}$

$F_w > F_{eff}$

By substituting the considered value of gear teeth, module, face width, various parameters mentioned above are calculated and the design is thus safe according to the condition mentioned above.

Thus, the final details of various gears are as follows:

Table 2: Gear Details

Sr. No	Details	Sun	Planet	Ring
1.	Number of Teeth	19	17	53
2.	Module	2 mm	2 mm	2 mm
3.	Face-width	30mm	32 mm	30 mm

Fixing of Ring Gear

Fixing of a certain element is essential in Epicyclic Gearbox since it would not work as desired if all the elements are free to rotate. The ring gear has been fixed in this case. The holding torque i.e. the torque which is to be applied to prevent its motion is calculated by the following equation:

$T_{Input} + T_{Output} + T_{Holding} = 0$

Where,

$T_{holding}$ = Holding Torque (N-mm)

T_{Input} = Input Torque (N-mm)

T_{Output} = Output Torque (N-mm)

The ring gear is fixed by means of press fit into the casing. The required amount of press fit interference to ensure holding torque is calculated according to pressure vessel design as follows:

- $F_{HOlding} = T_{Holding}\ x\ R_{Contact}$

  Where,

  $F_{Holding}$ = Holding Force (N)

  $R_{contact}$ = Contact Radius (mm)

- $P = \frac{F_{Holding}}{\mu\ x\ A}$

  Where,

  P = Contact Pressure (N-mm)

  μ = Coefficient of friction between casing and gear

  A = Area of contact

- $\delta = P\ x\left[\frac{d}{\epsilon_0}\left[\frac{d_o{}^2 + d^2}{d_o{}^2 - d^2} + \upsilon_o\right] + \frac{d}{\epsilon_i}\left[\frac{d^2 + d_i{}^2}{d^2 - d_i{}^2} - \upsilon_i\right]\right]$

  Where,

  δ = Amount of interference required (mm)

  d = Contact Diameter (mm)

  ϵ_0 = Young's Modulus of Casing Material (GPa)

  d_o= Casing Outer Diameter (mm)

  υ_o= Poisson's Ratio of Casing Material

  ϵ_i = Young's Modulus of Gear Material (GPa)

  d_i = Gear Inner Diameter (mm)

  υ_i= Poisson's Ratio of Gear Material

Thus, by substituting material properties and various dimensions, the resultant interference is calculated as 50 micron. Hence while manufacturing, desired dimensions are achieved such that 50 micron interference fit exists between casing and ring gear. To reduce any chance of ring gear slip, three grub-screws through the casing into the ring gear are provided in addition to the press-fit.

DESIGN OF PLANET CARRIER AND PINS

In this system, Carrier Pins i.e. custom bolts are used which will be coupled to the carrier by means of tapping in the Carrier. On the shank portion of the pins, the planets will be free to rotate, courtesy of needle bearing press-fit into them. The shank length has been so adjusted that its length is slightly greater than the face-width of the planets to prevent exertion of undesirable forces on the planet due to excessive tightening. Brass washers are placed between the pins, planets

and carrier, planets to reduce the losses due to friction. In order to reduce any chance of loosening of pins in the carrier, locknuts are also fitted to the pins on the other side of the carrier.

The static analysis of carrier was carried out to check the safety of the design. Constraint was given to the shaft and tangential force on the planets was applied at area where the pins meet the carrier. Tetrahedral mesh was used and the number of nodes and elements was 132050 and 89539 respectively. The maximum stress produced was 543.35MPawhich is sufficiently less than the Yield strength of the material (750MPa) and the maximum deformation was 0.06952 mm and thus the carrier was safe. For the pin, the analysis was carried out to check whether the design is safe enough to withstand the tangential forces on the planet. Constraint was given to the area where they connect to the carrier and the tangential force of the planets was applied on the shank portion. Tetrahedral mesh was used and the number of nodes and elements was 88078 and 58609 respectively. The maximum stress was 530.06MPawhich is sufficiently less than the Yield Strength of the material (750MPa) and the maximum deformation was 0.092679 mm and thus the pins were safe.

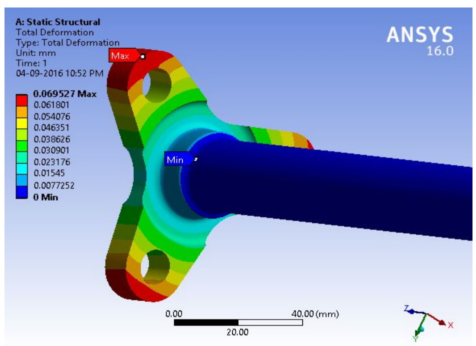


Figure 2: Carrier Analysis for Deformation

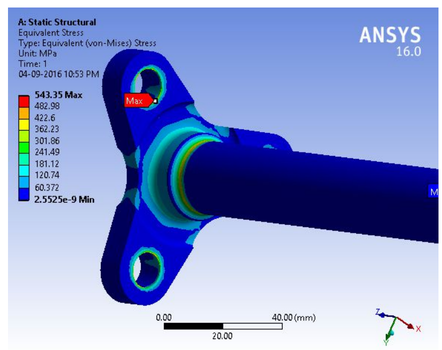


Figure 3: Carrier Analysis for Stress

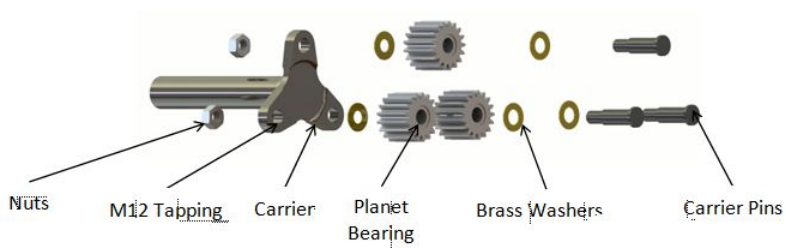


Figure 4: Carrier Exploded View

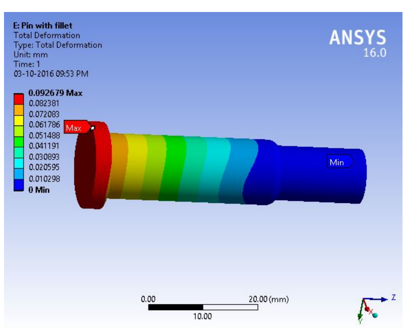


Figure 5: Pin Analysis for Deformation

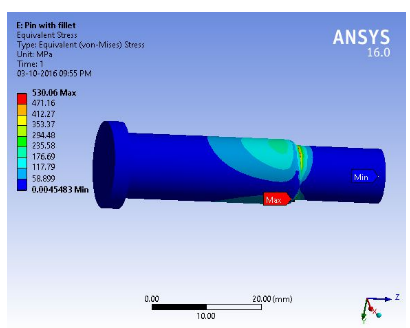
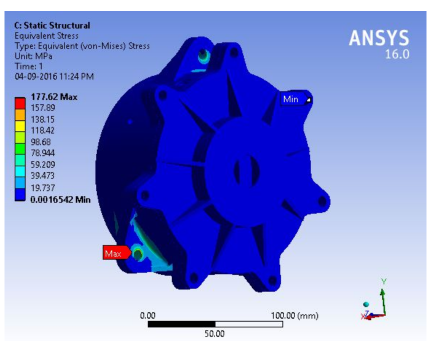


Figure 6: Pin Analysis for Stress

DESIGN OF CASING

The casing consists of two halves, each with a bearing press fit into it. The larger half has the ring gear press-fit into it and its bearing houses the sun gear, whereas the bearing in the smaller half houses the carrier shaft. The two halves are joined by means of seven bolting points. A step has been provided in the main casing half to realise the depth till which the ring gear has to be press-fit. To ensure co-axiality of grub-screw holes in the casing and the ring gear, a key way has been provided in both of them which are matched before fitting the ring gear into the casing. In order to achieve co-axiality of both the casing halves, a protrusion in the main half and a respective step in the other half have been provided. Three mounting points (one at the top and two at the bottom) are provided for fixing the gearbox on the chassis of the vehicle. Sufficient thickness along with axial ribs (Nine in number for effectively holding on three jaw chuck) is provided in areas adjacent to the bearing. Inlet and outlet ports are provided in the casing to enter and remove the oil respectively. A breather port has also been provided to prevent building of undesirable high air pressure. The material selected for the casing is Aluminium 7082 having tensile strength of 300MPa.

The static analysis of the casing (both the halves) was carried out to check the safety of the design. Constraint was given at the mounting points and the bearing forces were applied at their respective locations. Also, a torque equal in magnitude and opposite in direction to the holding torque is applied on the inner surface of the larger casing half since it houses the ring gear. Tetrahedral mesh was used and the number of nodes and elements was 1556093 and 1092480 respectively. The maximum stress was 177.62MPa which was sufficiently less than the Yield Strength of the material (300MPa) and the maximum deformation was 0.01752 mm, both of which rendered the casing safe.



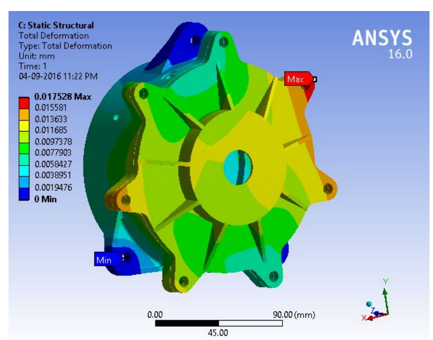


Figure 7: Casing Analysis for Deformation **Figure 8: Casing Analysis for Deformation**

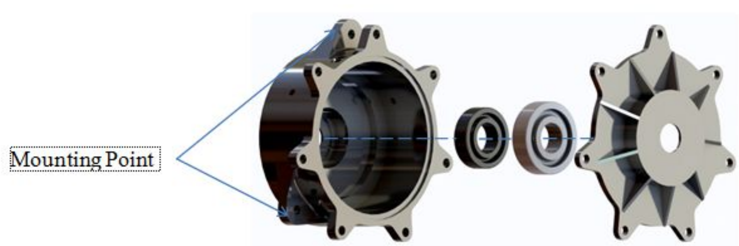


Figure 9: Casing Exploded View

Final Assembly

The final assembly consisting of all the parts is as follows

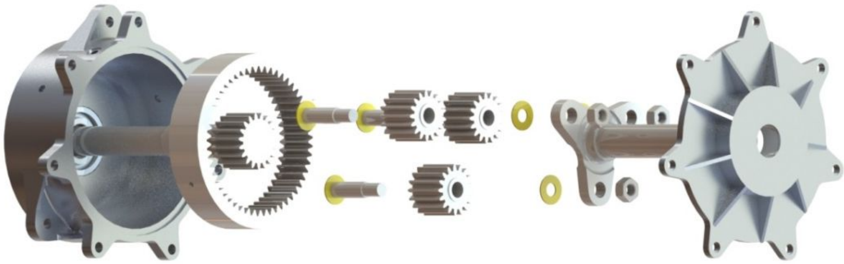


Figure 10: Gearbox Exploded View

MANUFACTURING AND ASSEMBLY

Sun and planet gears are manufactured by PG (Pre-Grinding) hobbing (providing economical and efficient operation), then case hardened by carburising and finished by gear grinding. The ring gear is manufactured by gear shaping, the carrier profile is obtained by gas cutting while the pins are manufactured on Lathe machine. All the above mentioned parts are case hardened by carburising to 58 HRC. The casing is manufactured by block machining. Two entire blocks of box dimensions of the two casing halves are taken and the casing halves are manufactured using turning on lathe and by means of VMC (Vertical Machining Centre). Thus intricate shape of the casing can be manufactured with good surface finish as well as sufficient strength.

First, the two bearings are press-fit into the respective casing halves and the needle bearings are press-fit into the planet gears. The pins after passing through the planets are bolted to the carrier and nuts are tightened on them to prevent loosening. The ring gear is press-fit into the casing half by matching the key ways in the ring and the casing and the grub-screws are tightened. The sun gear shaft is inserted into the main casing half. The carrier shaft is inserted into the main casing half where the planets mesh with the sun and the ring gear. The other half of the casing is inserted on the carrier shaft and the two halves are bolted. Oil (SAE 80W90) of quantity 200 ml is inserted into the casing through the oil inlet port and it can be drained after use through the oil outlet port provided at the bottom. The gearbox is now ready to use.

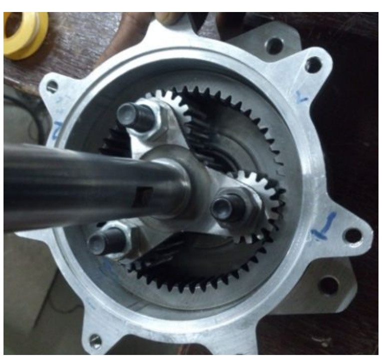


Figure 11: Gears and Carrier Prototype **Figure 12: Casing Prototype**

CONCLUSIONS

Therefore, the epicycle gearbox was designed, analysed and manufactured. Required amount of reduction (3.789) was achieved through it. It was tested on the All-Terrain Vehicle (ATV) successfully.

REFERENCES

1. *Design of Machine Elements, V.B. Bhandari (2012)*
2. *Handbook of Gear Design, GitinMaitra*
3. *Dudley's Handbook of Practical Gear Design and Manufacture*
4. *Handbook of Gear Technology, D.D Chawathe*
5. *Material Science and Metallurgy for Engineers, VD Kodgire*
6. *PSG Design Data-book, PSG College, Coimbatore*
7. *Shigley's Mechanical Engineering Design, Ninth Edition*